

Single-piston alternative to Stirling engines

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ARTICLE INFO

Article history:

Received 28 September 2011

Received in revised form 6 December 2011

Accepted 11 December 2011

Available online 12 January 2012

Keywords:

Energy
Stirling engine
Heat engine
Micro-CHP
Bio-oil

ABSTRACT

Thermodynamic analysis of an unconventional heat engine was performed. The engine studied has a number of advantages compared to state-of-the-art Stirling engines. The main advantage of the engine proposed is its simplicity. A power piston is integral with a displacer and a heat regenerator. It allows solving the problem of the high-temperature sealing of the piston and the displacer typical of all types of Stirling engines. In addition the design proposed provides ideal use of the displacer volume eliminating heat losses from outside gas circuit. Both strokes of the piston are working ones in contrary to any other types of piston engines. The engine can be considered as maintenance-free as it has no piston rings or any other rubbing components requiring lubrication. The only seal is contactless and wear free. It is located in the cold part of the cylinder. As a result the leakage rate in operation can be one-two orders of magnitude as small as that in Stirling engines. Balancing of the engine is much easier compared to Stirling engines with two reciprocating masses because of the only moving part inside the engine cylinder. The engine suits ideally to be fuelled with “difficult” fuels such as bio oil and can be used as a prime mover for micro-CHP systems.

The thermodynamic model developed incorporates non-ideal features of the cycle, such as specific regenerator efficiency, dead volumes and other geometrical parameters of the engine. The model shows that the energy efficiency is highly sensitive to regenerator performance. For realistic geometric and operating parameters and the regenerator efficiency of about 95% the ultimate energy conversion efficiency of the engine proposed can be as high as 40–50%.

A prototype of the engine was built and the feasibility of the engine concept was demonstrated.

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1. Introduction

The strength of external combustion heat engines vis-à-vis internal combustion engines is compatibility with a wide variety of renewable energy and fuel sources. They may use a supply of heat from any sources such as biomass and biomass derived products, municipal waste, nuclear, solar, and geothermal energy. Other important advantages of external combustion engines are low emissions due to continuous combustion and low noise, due to elimination of exhaust of high-pressure combustion products.

Promising external combustion engine concepts are Stirling engines that convert thermal energy into mechanical energy of reciprocating piston(s). The pistons are moved due to a cyclic change of gas phase working fluid pressure caused by its temperature and volume change. High thermal efficiency of the Stirling cycle (Carnot efficiency), long maintenance interval and fewer moving parts are additional advantages of Stirling engines [1].

In the operation of practical Stirling engines a significant deviation from the ideal cycle due to frictional losses, working fluid leakage, dead volumes, etc. takes place. Technical problems, in particular balancing of pistons (or piston and displacer) reciprocating with a phase lag and a high-temperature sealing of the piston, currently prevent the wide application of Stirling engines.

Significant improvements of Stirling engines are expected by using engines in which the displacer is integrated with the power piston. In these engines the piston–displacer assembly is moved due to the cycling pressure change of the working fluid under the influence of change of its temperature and amount inside the working chamber.

The purpose of this study was to perform thermodynamic analysis of one of such engines.

2. External combustion single-piston engine

The engine studied in this article closely resembles the hot-air engine devised by Manson [2] and the machines patented by Bush [3] and Baumgardner et al. [4]. Fig. 1 illustrates the basic principle of the engine cycle.

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point when the gas pressure is known and equal to the ambient pressure or the pressure in the engine casing P_0 .

$$P_0 = \frac{N_{uC}}{(h + x_2)S\beta} RT_C$$

$$P_0 = \frac{N_{uH}}{x_1 S} RT_H$$

$$N_u = N_{uC} + N_{uH} = \frac{P_0 S}{R} \left[\frac{(x_2 + h)\beta}{T_C} + \frac{x_1}{T_H} \right] = \frac{P_0 S h}{RT_C} ((1 + x'_2)\beta + x'_1 \alpha)$$

$$= N_0 ((1 + x'_2)\beta + x'_1 \alpha)$$

$$\text{where } N_0 = \frac{P_0 S h}{RT_C}; \quad x'_1 = \frac{x_1}{h}; \quad x'_2 = \frac{x_2}{h}.$$

3.1.2. Downward motion

Accordingly, when the piston moves down the mass of the gas in the cylinder can be calculated at the upper dead point.

$$P_0 = \frac{N_{dC}}{x_2 S \beta} RT_C$$

$$P_0 = \frac{N_{dH}}{(x_1 + h)S} RT_H$$

$$N_d = N_{dC} + N_{dH} = \frac{P_0 S}{R} \left[\frac{x_2 \beta}{T_C} + \frac{x_1 + h}{T_H} \right] = N_0 [x'_2 \beta + (1 + x'_1) \alpha]$$

The difference in the gas amount in the cylinder during upward and downward movements is

$$\Delta N = N_u - N_d = \frac{P_0 S h}{R} \left[\frac{\beta}{T_C} - \frac{1}{T_H} \right] = N_0 (\beta - \alpha)$$

One would expect the engine performs positive work if $\Delta N > 0$, or $T_H \beta > T_C$, or $\beta > \alpha$.

3.2. Gas pressure in the engine

When the displacer is in the position x the gas pressure as a function of x can be expressed as

$$P(x) = \frac{N_H}{S(x_1 + x)} RT_H = \frac{(N - N_H)}{S\beta(x_2 + h - x)} RT_C \quad (1)$$

where N is the amount of gas (number of moles) in the cylinder: $N = N_u$ – during the upward motion and $N = N_d$ – during the downward motion.

From Eq. (1) we have

$$N_H = \frac{NT_C}{(x_2 + h - x) \left[\frac{T_H \beta}{x_1 + x} + \frac{T_C}{(x_2 + h - x)} \right]}$$

Therefore

$$P(x) = \frac{NRT_C T_H}{(x_2 + h - x) \left[\frac{T_H \beta}{x_1 + x} + \frac{T_C}{(x_2 + h - x)} \right]} S(x_1 + x)$$

$$= \frac{NRT_C T_H}{S[T_H \beta(x_2 + h - x) + T_C(x_1 + x)]} = \frac{NA}{B - x}$$

where

$$A = \frac{RT_C T_H}{S[T_H \beta - T_C]} = \frac{RT_C}{S[\beta - \alpha]}; \quad B = \frac{h[\beta(1 + x'_2) + \alpha x'_1]}{\beta - \alpha}.$$

3.3. Mechanical work performed by the piston and engine power

During the upward stroke the work done is

$$W_u = S(1 - \beta) \int_0^h (P_u(x) - P_0) dx = S(1 - \beta) \left[N_u \int_0^h \frac{A}{B - x} dx - P_0 h \right]$$

$$= S(1 - \beta) \left[N_u A \ln \frac{B}{B - h} - P_0 h \right]$$

During the downward stroke the work done is

$$W_d = S(1 - \beta) \int_h^0 (P_d(x) - P_0) dx = S(1 - \beta) \left[N_d \int_h^0 \frac{A}{B - x} dx + P_0 h \right]$$

$$= S(1 - \beta) \left[-N_d A \ln \frac{B}{B - h} + P_0 h \right]$$

The total useful work per cycle is

$$W = W_u + W_d = S(1 - \beta)(N_u - N_d)A \ln \frac{B}{B - h} = S(1 - \beta)\Delta N A \ln \frac{B}{B - h}$$

Using the expressions for ΔN , A , and B we get

$$W = P_0 h S(1 - \beta) \ln \frac{T_H \beta(x_2 + h) + T_C x_1}{T_H \beta x_2 + T_C(x_1 + h)} = P_0 h S(1 - \beta) \ln \frac{\frac{\beta}{\alpha}(1 + x'_2) + x'_1}{\frac{\beta}{\alpha}x'_2 + 1 + x'_1}$$

The work done can be rewritten as

$$W = P_0 h S(1 - \beta) \ln \left(1 + \frac{\frac{\beta}{\alpha} - 1}{1 + \frac{\beta}{\alpha}x'_2 + x'_1} \right)$$

It shows that the maximum mechanical work is achieved if $x_1 = x_2 = 0$. In this case

$$W = P_0 h S(1 - \beta) \ln \frac{\beta}{\alpha}; \quad W' = \frac{W}{P_0 h S} = (1 - \beta) \ln \frac{\beta}{\alpha}$$

If f is the frequency of the reciprocation then the mechanical power of the engine is

$$P_W = f P_0 h S(1 - \beta) \ln \frac{\beta}{\alpha}. \quad (2)$$

4. Energy efficiency

The cycle is composed of four stages:

1. The displacer moves up. The volume of the hot space increases from Sx_1 to $S(x_1 + h)$. The amount of the gas in the hot space increases accordingly.
2. The gas discharges from the cylinder. The piston/displacer is in the upper dead point. The volume of the hot space remains constant. The pressure drops to P_0 , the amount of the gas decreases.
3. The displacer moves down. The volume of the hot space decreases from $S(x_1 + h)$ to Sx_1 . The amount of the gas in the hot space decreases accordingly.
4. The gas comes in/intakes to the cylinder. The piston/displacer is in the lower dead point. The volume of the hot space remains constant. The pressure rises to P_0 , the amount of the gas increases.

The general energy conservation equation and equation of the gas state are

$$\delta Q_T = dU + PdV; \quad T = \frac{PV}{NR}$$

where $U = NC_V T$ – internal energy; N – number of moles; Q_T is the total amount of heat coming into the system (hot space). The temperature of the gas in the hot space is T_H .

4.1. Stage 1

The energy conservation equation for the gas in the hot space of the engine can be rewritten as

$$Q_{1T} = \Delta N_{H1} C_p T_{1in} + Q_1 = C_v T_H \Delta N_{H1} + S \int_0^h P(x) dx \quad (3)$$

where ΔN_{H1} the number of moles coming into the hot space, or the increase/change in the number of moles in the hot space during the 1st stage; Q_1 – the amount of heat transferred to the hot space during stage 1; T_{1in} – the temperature of the gas coming into the hot space. It depends on properties of the regenerator and heat exchange between the gas and regenerator.

In the ideal case the heat capacity of the regenerator (displacer) is much larger than the total heat capacity of the gas in the cylinder, the heat exchange between the gas and the regenerator is instantaneous, the thermal conductivity of the regenerator along its axis is zero, and the temperature of the regenerator changes along its axis from T_H to T_C .

Modeling of the heat exchange between the gas and displacer/regenerator requires special consideration. Here we use simplified approach.

The temperature of the gas coming into the hot space will be in between T_C and T_H

$$T_{1in} = r_1 T_H + (1 - r_1) T_C$$

where r can be called the regeneration efficiency ($0 \leq r \leq 1$).

From Eq. (3) and taking into account the calculations of the work above we get

$$\begin{aligned} Q_1 &= -\Delta N_{H1} (C_p T_{1in} - C_v T_H) + S N_u A \ln \frac{B}{B-h} \\ &= -\Delta N_{H1} ((C_p r_1 - C_v) T_H + C_p (1 - r_1) T_C) + S N_u A \ln \frac{B}{B-h} \end{aligned}$$

$$\Delta N_{H1} = N_{H1f} - N_{H1i}$$

where ΔN_{H1} is the change/increase in number of moles in the hot space during stage 1; N_{H1f} and N_{H1i} are the numbers of moles in the hot space in the end and in the beginning of stage 1. From the equations expressing the pressures in the hot and cold parts

$$\frac{T_H N_{H1i}}{x_1} = \frac{T_C (N_u - N_{H1i})}{(x_2 + h)\beta}, \quad \frac{T_H N_{H1f}}{x_1 + h} = \frac{T_C (N_u - N_{H1f})}{x_2 \beta}$$

we find

$$\Delta N_{H1} = N_u \left[\frac{1 + x'_1}{1 + x'_1 + x'_2 A} - \frac{x'_1}{(1 + x'_2) A + x'_1} \right]$$

where

$$A = \beta/\alpha.$$

During stage 1 the regenerator will get the heat (basically take off the heat)

$$Q_{r1} = -\Delta N_{H1} C_p (T_{1in} - T_C) = -\Delta N_{H1} C_p r_1 (T_H - T_C)$$

If the dead volume is neglected ($x_1 = x_2 = 0$)

$$\Delta N_{H1} = N_u = \frac{P_0 S h \beta}{R T_C} = N_0 \beta$$

and

$$Q_1 = -N_0 \beta ((C_p r_1 - C_v) T_H + C_p (1 - r_1) T_C) + S N_0 \beta A \ln A$$

$$Q_{r1} = -N_0 \beta r_1 (T_H - T_C).$$

4.2. Stage 2

The energy conservation equation for the gas in the hot part during the gas release from the cylinder is

$$Q_{2T} = -\Delta N_{H2} C_p T_H + Q_2 = \int dU = -\Delta N_{H2} C_v T_H$$

or

$$Q_2 = \Delta N_{H2} R T_H$$

$$\Delta N_{H2} = N_{H2i} - N_{H2f}$$

where ΔN_{H2} is the change/decrease in number of moles in the hot space during stage 2; N_{H2f} and N_{H2i} are the numbers of moles in the hot space in the end and in the beginning of stage 2.

The decrease in the number of moles in the hot space during the gas discharge can be found from the equations:

$$\frac{T_H N_{H2i}}{x_1 + h} = \frac{T_C (N_u - N_{H2i})}{x_2 \beta}, \quad \frac{T_H N_{H2f}}{x_1 + h} = \frac{T_C (N_d - N_{H2f})}{x_2 \beta}$$

As a result

$$\Delta N_{H2} = (N_u - N_d) \frac{1 + x'_1}{1 + x'_1 + x'_2 A} = \Delta N \frac{1 + x'_1}{1 + x'_1 + x'_2 A}$$

Therefore, the amount of the heat supplied to the hot space during stage 2 is

$$Q_2 = \frac{\Delta N (1 + x'_1) R T_H}{1 + x'_1 + x'_2 A}; \quad \Delta N = N_0 (\beta - \alpha)$$

The regenerator will receive during stage 2 the heat

$$\begin{aligned} Q_{r2} &= \Delta N_{H2} C_p (T_H - T_{2out}) = \Delta N_{H2} C_p (T_H - (r_2 T_C + (1 - r_2) T_H)) \\ &= \Delta N_{H2} C_p r_2 (T_H - T_C) \end{aligned}$$

where $T_{2out} = r_2 T_C + (1 - r_2) T_H$ is the temperature of the gas coming into the cold space. If the dead volume is neglected

$$Q_2 = N_0 (\beta - \alpha) R T_H$$

$$Q_{r2} = N_0 (\beta - \alpha) C_p r_2 (T_H - T_C).$$

4.3. Stage 3

The energy conservation equation for the gas in the hot space of the engine during downward movement of the displacer can be rewritten as

$$Q_{3T} = -\Delta N_{H3} C_p T_H + Q_3 = -C_v T_H \Delta N_{H3} + S \int_h^0 P(x) dx$$

$$Q_3 = R T_H \Delta N_{H3} - S N_d A \ln \frac{B}{B-h}$$

$$\Delta N_{H3} = N_{H3i} - N_{H3f}$$

where ΔN_{H3} is the decrease in number of moles in the hot space during stage 3; N_{H3i} and N_{H3f} are the numbers of moles in the hot space in the beginning and in the end of stage 3. From the equations

$$\frac{T_H N_{H3i}}{x_1 + h} = \frac{T_C (N_d - N_{H3i})}{x_2 \beta}, \quad \frac{T_H N_{H3f}}{x_1} = \frac{T_C (N_d - N_{H3f})}{(x_2 + h) \beta}$$

we find

$$\Delta N_{H3} = N_d \left[\frac{1 + x'_1}{1 + x'_1 + A x'_2} - \frac{x'_1}{A (1 + x'_2) + x'_1} \right]$$

The regenerator will receive during stage 3 the heat

$$\begin{aligned} Q_{r3} &= \Delta N_{H3} C_p (T_H - T_{3out}) = \Delta N_{H3} C_p (T_H - (r_3 T_C + (1 - r_3) T_H)) \\ &= \Delta N_{H3} C_p r_3 (T_H - T_C) \end{aligned}$$

where $T_{3out} = r_3 T_C + (1 - r_3) T_H$ is the temperature of the gas coming into the cold space.

If the dead volume is neglected

$$\Delta N_{H3} = N_d = \frac{P_0 S h}{RT_H} = N_0 \alpha$$

and

$$Q_3 = RT_H N_0 \alpha - \alpha S N_0 A \ln A$$

$$Q_{r3} = \alpha N_0 C_p r_3 (T_H - T_C).$$

4.4. Stage 4

$$Q_{4T} = \Delta N_{H4} C_p T_{4in} + Q_4 = \int dU = \Delta N_{H4} C_v T_H$$

where $T_{4in} = r_4 T_H + (1 - r_4) T_C$ is the temperature of the gas coming into the hot space.

$$Q_4 = -\Delta N_{H4} (C_p T_{4in} - C_v T_H) = -\Delta N_{H4} ((C_p r_4 - C_v) T_H + C_p (1 - r_4) T_C)$$

$$\Delta N_{H4} = N_{H4f} - N_{H4i}$$

The increase in the number of moles in the hot space during the gas intake can be found from the equations:

$$\frac{T_H N_{H4i}}{x_1} = \frac{T_C (N_d - N_{H4i})}{(x_2 + h)\beta}, \quad \frac{T_H N_{H4f}}{x_1} = \frac{T_C (N_u - N_{H4f})}{(x_2 + h)\beta}$$

As a result

$$\Delta N_{H4} = (N_u - N_d) \frac{x'_1}{x'_1 + (1 + x'_2)A} = \frac{\Delta N x'_1}{x'_1 + (1 + x'_2)A}$$

Therefore, the amount of the heat supplied to the hot space during stage 4 is

$$Q_4 = -\frac{\Delta N x'_1}{x'_1 + (1 + x'_2)A} ((C_p r_4 - C_v) T_H + C_p (1 - r_4) T_C)$$

The regenerator will receive (get off) during stage 4 the heat

$$Q_{r4} = -\Delta N_{H4} C_p (T_{4in} - T_C) = -\Delta N_{H4} C_p (r_4 T_H + (1 - r_4) T_C - T_C) = -\Delta N_{H4} C_p r_4 (T_H - T_C)$$

If the dead volume is neglected

$$Q_4 = 0 \quad \text{and} \quad Q_{r4} = 0$$

The amount of gas in the hot space at the end of the cycle is the same as at the beginning of the cycle. Therefore

$$\Delta N_{H1} - \Delta N_{H2} - \Delta N_{H3} + \Delta N_{H4} = 0$$

The total amount of heat supplied to the regenerator is

$$Q_{rT} = \sum_{i=1}^{i=4} Q_{ri} = -\Delta N_{H1} C_p r_1 (T_H - T_C) + \Delta N_{H2} C_p r_2 (T_H - T_C) + \Delta N_{H3} C_p r_3 (T_H - T_C) - \Delta N_{H4} C_p r_4 (T_H - T_C) = -C_p (T_H - T_C) (\Delta N_{H1} r_1 - \Delta N_{H2} r_2 - \Delta N_{H3} r_3 + \Delta N_{H4} r_4)$$

Since $Q_{rT} = 0$ we may assume that $r_1 = r_2 = r_3 = r_4 = r$.

For simplicity we assume that the dead volume is zero. In this case the total amount of heat supplied to the hot space per cycle is

$$Q_T = \sum_{i=1}^{i=4} Q_i = -N_0 \beta ((C_p r - C_v) T_H + C_p (1 - r) T_C) + S N_0 \beta A \ln A + N_0 (\beta - \alpha) R T_H + R T_H N_0 \alpha - \alpha S N_0 A \ln A$$

or

$$Q_T = -N_0 \beta ((C_p r - C_v - R) T_H + C_p (1 - r) T_C) + S N_0 (\beta - \alpha) A \ln A = N_0 \beta C_p (1 - r) (T_H - T_C) + S N_0 (\beta - \alpha) A \ln A = \frac{P_0 S h}{RT_C} \left[\beta C_p T_H (1 - r) (1 - \alpha) + S (\beta - \alpha) \frac{RT_C}{S(\beta - \alpha)} \ln A \right] = P_0 S h \left[\frac{\beta}{\alpha \gamma - 1} (1 - r) (1 - \alpha) + \ln A \right]$$

Finally the thermodynamic efficiency of the engine is

$$\eta_0 = \frac{W}{Q_T} = \frac{(1 - \beta) \ln \frac{\beta}{\alpha}}{\frac{\beta}{\alpha} \frac{\gamma}{\gamma - 1} (1 - r) (1 - \alpha) + \ln \frac{\beta}{\alpha}}. \quad (4)$$

5. Discussion

The study performed shows that energy efficiency of the engine is determined by: (1) the properties of the working fluid; (2) the ratio of the diameters of the displacer and piston; (3) the ratio of the temperatures of the heater and cooler; (4) the efficiency of the heat regenerator.

Figs. 3 and 4 show examples of the calculated thermodynamic efficiencies as a function of the ratio of the areas of the displacer S and piston S_p .

The model shows that thermal efficiency is highly sensitive to regenerator performance. In case of ideal regenerator ($r = 1$) the efficiency is

$$\eta_0 = 1 - \beta = \frac{S_p}{S}$$

Since the engine produces power if $\beta > \alpha$ the thermodynamic efficiency is always less than the ideal efficiency. It approaches the

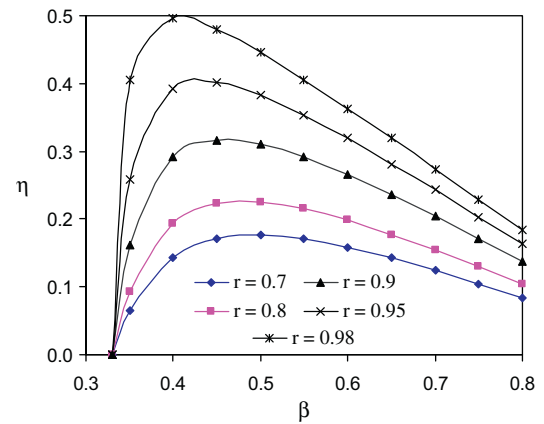


Fig. 3. Thermodynamic efficiency of the engine as a function of the geometrical parameter $\beta = (S - S_p)/S$ for different regeneration efficiencies and $\alpha = T_C/T_H = 0.33$.

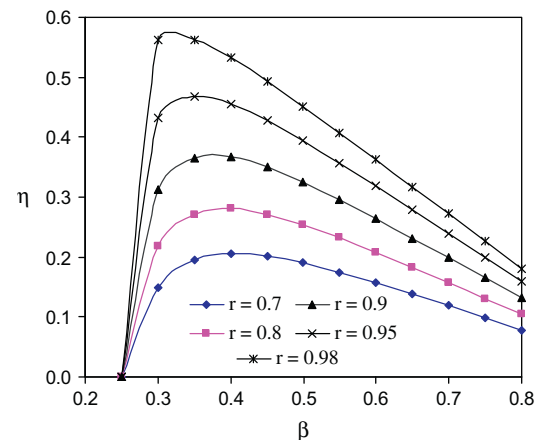


Fig. 4. Thermodynamic efficiency of the engine as a function of the geometrical parameter $\beta = (S - S_p)/S$ for different regeneration efficiencies and $\alpha = T_C/T_H = 0.25$.

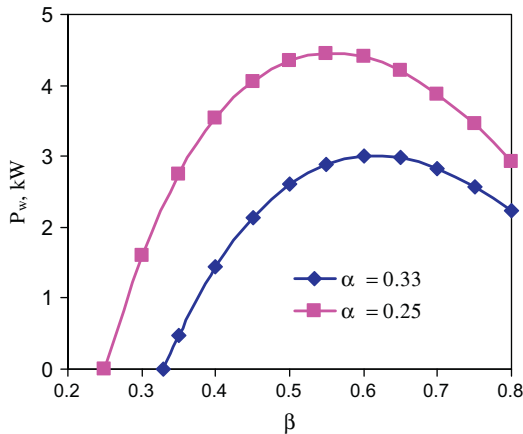


Fig. 5. Mechanical power of the engine as a function of the geometrical parameter $\beta = (S - S_p)/S$ for different temperature ratios $\alpha = T_C/T_H = 0.25$ and 0.33 .

ideal efficiency if β tends to a limit. However the engine power in this case tends to zero (Eq. (2)).

Although the engine efficiency is always less than Carnot efficiency it can be as high as 50% for realistic geometric and operating parameters and the regenerator efficiency of about 95%. Modeling of the heat regenerator and its optimization showed that its efficiency of about 95% can easily be achieved. Moreover, it can be made very close to 100%, say 98%. In that case the efficiency of the engine can be as high as 50–60% provided the geometrical parameter $\beta = (S - S_p)/S$ has appropriate value.

In view of its advantages energy losses in the engine are expected to be much less than in Stirling engines. To estimate the influence of possible energy losses on the energy efficiency a detailed modeling of the engine was performed (to be reported). Due to difficulties with the calculations of the gas flow between the hot and cold parts of the engine the minimum temperature ratio $\alpha = T_C/T_H$ was 0.54. The results showed that the difference between the obtained thermodynamic efficiency, Eq. (4), and the expected practical efficiency is not dramatic. For example, at the temperature ratio of 0.54 the efficiency can be as high as 30%.

Fig. 5 shows an example of the calculated power (Eq. (2)) for temperature ratios $\alpha = T_C/T_H = 0.25$ and 0.33 as a function of the

ratio of the areas of the displacer S and piston S_p . The power was calculated for the following values of the engine parameters: inner diameter – 40 mm, piston–displacer stroke – 50 mm, pressure of the working fluid – 10^7 Pa, $\gamma = 1.667$, $f = 20$ Hz.

A prototype of the engine coupled with crank gear was built and the feasibility of the engine concept was demonstrated [5]. The inner diameter of the engine cylinder is 20 mm and the piston–displacer stroke is also 20 mm. The tests of the engine are under way.

6. Conclusions

The results show that optimal design and operating parameters should be selected as a compromise between power and efficiency. Very compact engine is capable of producing 2–4 kW power with practical efficiency above 30%.

The engine studied has a number of advantages compared to state-of-the-art Stirling engines and suits ideally to be fuelled with “difficult” low-grade fuels such as bio-oil and can be used as a prime mover for micro-CHP systems. It seems to be an effective alternative and supplement to the established methods of energy conversion: internal combustion engines, microturbines, Stirling engines and fuel cells.

Acknowledgements

This study has been carried out within the framework of the first Russian Federation–European Union cooperative project “Bio-liquids-CHP”, co-funded under the FP7 scheme from the European Commission (EC) for the EU-members and the Federal Agency for Science and Innovation (FASI) of the Russian Federation for the Russian partners.

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